THE PULSE JET - A PROPULSIVE DEVICE
FOR DEEP OCEAN VEHICLES

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ABSTRACT

The intent of this paper is to provide an initial study of an underwater pulse jet with a view toward its use in deep submergence vehicles.

A simplified model of the pulse jet is presented so that the influence of the operating variables (depth, maximum power requirement, and endurance) on the design parameter (stroke, geometry, gas pressure, size and cycle time) may be determined. The value of this model is limited by the fact that the governing equations are nonlinear and that an analytical solution is not possible. In addition, general comments on the influence of losses, gas generation, interface phenomena, and exhaust phenomenon are presented.

On the basis of this research, it is the writer's opinion that the immediate use of the pulse jet in a deep ocean environment is not recommended because its fuel consumption increases with depth, and because no use can be made of gas expansion at great depths. However, for shallow applications the pulse jet may prove to be quite feasible.

While pulsedoperation was achieved, an attempted verification of a mathematical model was not completely successful due to insufficient pressure regulation in the experiment, as described herein.

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NOMENCLATURE

ρ	-	sea water density
c	-	water ejection velocity
v_∞	-	vessel speed or water inlet velocity
p	-	pressure
t	-	time
h	-	piston position
A	-	cross sectional area
x, y	-	horizontal and vertical reference axes, respectively
B, C, D	-	constants of integration
L	-	length as defined
R	-	gas constant
S	-	stroke
T	-	thrust
F	-	force
g	-	acceleration of gravity
c.v.	-	control volume

Subscripts

a	-	ambient
g	-	gas
y,0,1,2,3	-	at sections y,0,1,2,3 as respectively defined
D, E, & I	-	referring to the drift, ejection, and inlet phases, respectively

MOTIVATION

The awakening of the scientific community to the potential of the undersea world has led to a recent rekindling of interest in ocean engineering. One area of prime interest is that of underwater propulsion. While both shallow and deep operating vehicles have been driven by screw propellers quite satisfactorily for many years, the recent effort to explore the depths has led to new constraints and a re-examination of the propulsion schemes for deep submersibles.

At this point one might ask: "Why won't propellers and associated equipment do here?" The answer is that they will, but other devices might perform more satisfactorily. By defining what is meant by more satisfactory, we shall set a criterion for optimum design.

As the depth requirement of a submarine increases, the hull weight as a fraction of displacement also increases. Thus, to provide for a maximum scientific payload weight, other submarine systems' weights must be kept to a minimum. In the case of the propulsion system, this can be done by mounting a minimum weight plant external to the pressure hull, thereby reducing its effective weight by the bouyant force of the water displaced.

Thus, we shall determine the optimum propulsion system to be of minimum weight mounted external to the pressure hull as determined by the weight limitations.

Simplicity of concept is another factor which we will determine as desirable in the early stages of power plant development.

While simplicity at later stages relates back to other design parameters, few will disagree with the desirability of early concept simplicity.

In this undertaking, the single most important criterion for selection of a means of energy storage is maximum storable energy per unit weight. As the chemical storage devices have the highest energy per unit weight, they will be the type of energy storage selected for this weight-limited application and will be the only type of devices considered in further discussions.

In a water environment, thrust is derived by increasing the momentum of a portion of that fluid. The problem in developing a propulsion system for this application then appears to be finding a means to convert the potential energy of the fuel to an increase in kinetic energy of the water.

An immediately obvious solution is to allow the relatively high pressure products of combustion to expand against the water in an open cycle. In this manner, a multitude of mechanical, electrical, and/or thermal energy conversion and transfer devices may be eliminated. Note also that this type of device operating on an open cycle and having no working fluid is not limited by the Carnot efficiency.

By using minimum weight, a mounting extermal to the pressure hull requirement, and concept simplicity as primary preliminary considerations in the development of a deep submarine power plant, we shall explore the possibility of using a direct energy conversion device as a means of propulsion.

For the propulsion system designer, the high pressure sea

water environment provides advantages in that he has at hand a relatively constant temperature heat sink and the possibility of high heat transfer rates to this sink. However, he is also plagued with the following disadvantages:

1. The high ambient pressure requires some parts to be designed as pressure vessels.
2. Considerably more effort must be exerted to get rid of exhaust products.
3. There is corrosion of seawater, especially at high pressure.
4. Marine plants and animals contribute to the incrustation and corrosion of the exposed metal.
5. Very deep installations cannot be worked on by man from the sea side.
6. The power system may affect the ecology of the local sea to the detriment of the system.

Major design parameters to be considered in selecting a candidate propulsive device are: maximum operating depth, the endurance required or the projected mission time, and the total power requirements. Once these requirements are met, other factors to be taken into consideration are:

1. weight, size and/or shape limitations
2. reliability
3. safety
4. life-costs
5. shelf and useful life
6. recharging time
7. necessity to protect system from environment

8. support requirements
9. availability
10. state of the art
11. operating characteristics (i.e. effect of sea motion on controlability.)
12. maintainability

Before looking into the possibilities for a direct conversion device for small submarines, as a background we shall review those devices currently proposed for underwater applications. As one might suspect, some of these devices, such as the rocket, ram jet, and turbojet, have been developed long ago for use in air-borne vehicles. In general, these propulsive systems are characterized by high combustion product exhaust velocities and high fuel consumption rates, thus limiting their usefulness now to relatively high speed, short-run vehicles.

Noyes, in reference (3), has suggested using these devices in a fully cavitating missile. By designing for fully cavitating operation, he notes that the engineer has at his disposal a captive environment in which the power plant is to operate, that of low pressure, low velocity steam. Beneficial effects produced by operating in the environment are desensitization of the engine to depth and velocity, velocity change with constant thrust, and reduction of propulsion requirements.

The usefulness of rockets is limited by their very short operating time, about 240 sec maximum. Thus, it would appear that the only immediately feasible application of rockets would be in torpedo-like devices with a very short operating time.

The ram jet which operates by utilizing on board propellent combustion in connection with water taken in, also has a limiting characteristic--its operating time of only about 1200 seconds. Another critical aspect of the ram jet's operation is that its velocity must be initially boosted so that water is introduced into the combustion chamber at sufficient ram pressure to sustain operation. Without going into great detail, Hacker and Lieberman in reference (5) suggest other operating characteristics of a ram jet, as: a higher efficiency than a rocket, very noisy operation, narrow operating limits with regard to velocity of the vehicle, and best operating conditions at high velocity and at shallow depths. The above authors also note: "Thermodynamic analysis has shown that the underwater ram jet may never develop satisfactory thrust to overcome drag effects, especially with the propellents available today and when the possible energy losses are considered."

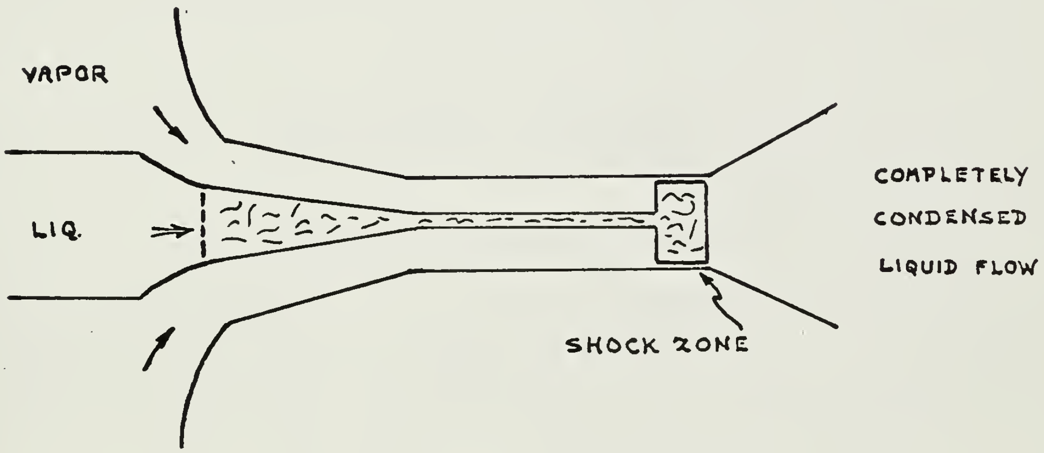
The turbojet cannot strictly be called a direct conversion device; however, we will consider it here, since it derives a portion of its thrust in this manner. Its maximum operating time is the greatest of the three, about 4000 sec; and, while it is more complex, it potentially offers the highest possible efficiency. Noyes notes that the turbojet-powered missile is comparable in range to the conventional torpedo.

A condensing ejector, while not a direct conversion device, is a mechanism by which high combustion product energy can be transferred to the sea water (See Figure I). A particular application for condenser ejectors as noted by Miguel and Brown

in reference (2) is for deep running torpedoes with an open-cycle turbine. A condensing ejector can be used to increase the relatively low pressure of the turbine gas exhaust to the high environmental pressure.

Figure I

Condensing Ejector



In the condensing ejector, the momentum of the liquid steam is increased as a result of condensation of the relatively high velocity vapor. Appreciable condensation is possible because of high velocity differences between the vapor and liquid steam which produce extremely high heat transfer coefficients.

Another means by which energy may be transferred from relatively high energy combustion products to the environment is through the use of "pseudo blades." In this device, as explained by Sarro and Avellone in reference (4), mechanical

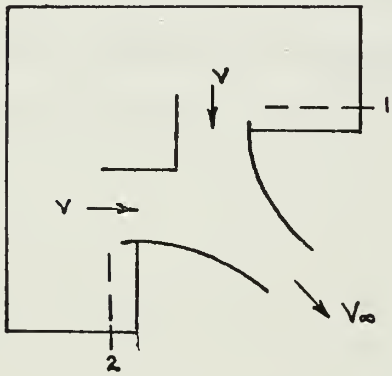
energy is transferred directly from one fluid to another with an ideal efficiency equal to that of rotating machinery (See Figure II). This device, by allowing a nondissappative interaction of two flow streams of equal velocity, one in a moving reference system, enables energy to be transferred between the streams when the velocities are referred to an absolute reference frame.

in relative frame $v_1 = v_2 = v$

in absolute frame $v_1 = 0, v_2 = v$

Figure II

Pseudo Blade Diagram



After interaction, both fluids have a common orientation and equal velocities in both frames. Thus, we see that energy has been exchanged between flows.

Sarro and Avellone suggest that a steam-water pseudo blade system might be suitable as a propulsor for torpedoes or other water-borne vehicles because of its simplicity, light weight,

efficiency at high speeds, and lack of visible wake. Furthermore, the overall efficiencies may approach 73%.

We have seen that some of the above direct conversion propulsion devices are used in high speed, torpedo-like applications. However, can a direct conversion system be used in propelling a research submarine, a larger slower speed vehicle of longer mission time? Before making a suggestion along these lines, we must determine more exactly the parameters of the vessel we are to consider.

In terms of the major propulsion design parameters, the vessel to be considered will be of 20,000 foot operating depth, (sufficient to reach 98% of the ocean bottom), capable of 10 hours endurance (the human limit with no living facilities on board), and of 20 hp(about double the propulsive power requirements of the small research vessel, Alvin, for 3 kt operation).

PULSE JET MODEL

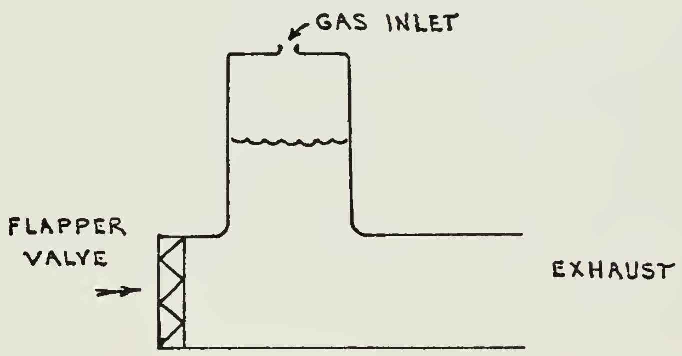
Concept

Recalling the primary requirements for the deep submarine power plant (operation external to the pressure hull and simplicity of design while providing sufficient power and endurance at the required depth), we now ask how might we most easily allow products of combustion to expand against the environmental water in an open cycle. The devices mentioned previously are an answer for high speed, torpedo-like applications. But here we are considering a larger, slower, not as hydrodynamically faired vessel whose operating characteristics differ so much from those of the previously mentioned vehicles that propulsion efficiencies would be very small to the point of rendering their devices useless.

The easiest way to expand against the water is to allow gas to expand directly against the water surface without intermediate transmission devices.

Figure III

Pulse Jet, Conceptual Drawing



Through the use of the device in Figure III, an intermittent thrust can be provided by operating on the following cycle:

1. Sea water is taken in through the flapper (i.e. check valve),
2. High pressure gas is generated,
3. The gas forces the flapper valve closed and the sea water out the exhaust.

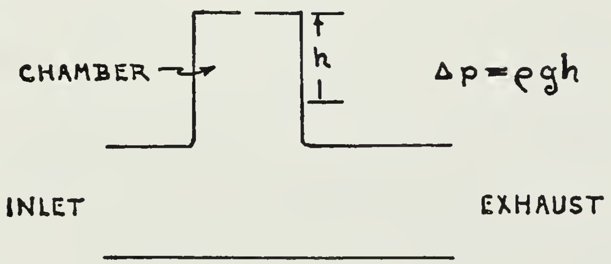
This device we shall refer to as a pulse jet, although calling it a water pump would be just as descriptive.

To be considered in turn will be the pulse jet characteristics and design considerations, plus specific comments on the influence of losses, gas generation, the gas-liquid interface, and sea water flow phenomenon.

Pulse Jet Characteristics

Before establishing the pulse jet design parameters, we must postulate a means of gas exhaust. A very simple method would be to let it exhaust to the environment through a valve at the top of the chamber, the head of sea water displaced being the driving force (See Figure IV).

Figure IV



Now consider the effect of the operating variables (depth, power requirement, and endurance), on the pulse jet design parameters. Obviously, operating at great depths will require increasing high absolute gas pressures and, as previously noted, will negate the tendency for gas expansion. The power requirement deserves further explanation in that in addition to the total power required we need to know the speed vs power curve of the particular hull being considered in order to determine the thrust required at various speeds. Endurance requirements at a particular depth will influence design parameters such that maximum efficiency is achieved at the endurance speed.

We shall determine the pulse jet design parameters to be:

1. size (cross sectional area)

2. geometry

3. gas pressure

4. stroke

5. cycle time

To avoid unnecessary complications at this early stage of development, we shall limit our discussion to a pulse jet of tee-shaped geometry with uniform inlet, chamber, and exhaust cross sectional areas, with no frictional losses, propelling a vessel at a constant speed.

As a consequence of operating at great depths (20,000 ft.), we shall consider the gas pressure during the sea water ejection or positive thrust phase to be constant.

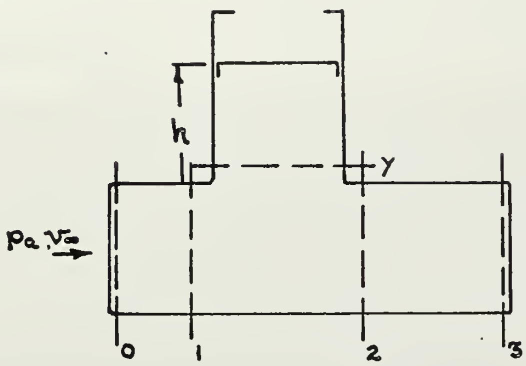
Furthermore, our considerations shall be limited to a pulse jet with a piston to eliminate the gas-liquid interface. To avoid additional complications, we shall consider the piston mass-less.

Let us now determine the pulse jet parameters for a particular application, as defined by the maximum speed required, the speed-power curve, and the size limitations.

First note that for positive time average to be developed by this intermittent device, the sea water ejection velocity must be greater than the inlet velocity. Furthermore, as thrust is intermittently positive and negative during the cycle, we must have knowledge of not only the thrust magnitudes, but of the relative duration of the phases in determining the time average thrust.

The cross section shown in Figure V will be used in studying the pulse jet cycle.

Figure V



The "ejection phase" begins with the closing of the exhaust valve and the injection of high pressure gas. The drift between the inlet and ejection phases will be neglected, as the much higher driving force in this case will result in an infinitesimal time being spent in the drift phase. The completion of the ejection phase coincides with the closing of the gas inlet valve.

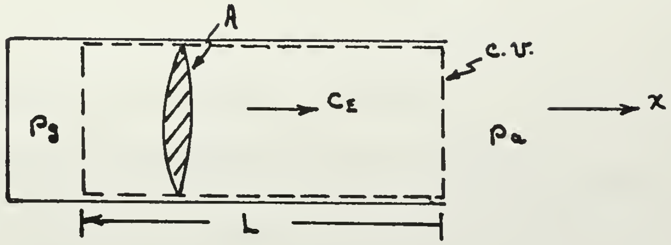
At the point in the cycle where the gas inlet injection process ends(the opening of the exhaust valve and closing of the gas inlet valve), water in the chamber continues to move downward. One would think that the water would be drawn upward into the chamber at this point, but due to inertia of the relatively high velocity mass of water in section 2-3, this water is drawn downward until possibly a piston stop is reached. After the stop is hit, the pressure upstream of section 2 decreases, causing an acceleration of the flow in section 0-1 and causing a deceleration of the flow in section 2-3. As the velocity along 0-3 decays (to v_∞) the pressure at y finally increases to the free stream pressure and the piston will begin to travel upward.

We shall call the above phenomenon the "drift phase."

The upward movement of the piston constitutes the "inlet phase." As the piston accelerates upward, it tends to decrease the momentum of the water in section 2-3 and to increase the momentum in section 0-1.

Cycle Analysis

Our examination of the ejection phase may be simplified by considering the control volume, c.v.



where L is the length of water accelerated, the stroke plus the downstream length.

The thrust on this device is just the result of the pressure difference, $p_g - p_a$, and is constant at a specified depth and at a constant gas pressure, p_g.

$$T_E = (p_g - p_a)A \tag{1}$$

But to determine the time average thrust we also must know the duration of this phase as a function of the pulse jet parameters. For a given stroke and geometry we then require the average ejection velocity. By writing a momentum balance in the x direction, an expression for entrained water length as a function of time is obtained, (equation (2)).(See Appendix A for derivation).

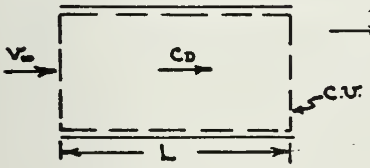
$$L \frac{d^2L}{dt^2} = -\left(\frac{p_g - p_a}{\rho}\right) \tag{2}$$

Note, however, that the momentum equation is, for this simplified model, nonlinear. Thus, no analytical solution will be

possible and numerical methods will have to be used for each particular case.

In our investigation of drift phase phenomena we shall consider the decay of velocity in a streamtube where pipe friction has been neglected and where length, L, is equivalent to the sum of the upstream and downstream lengths, L_{0-3}.

Equation (3) is the simplified momentum balance for control volume, c.v., (See Appendix A for derivation).



$$L\frac{dc_D}{dt} + c_D^2 = v_\infty^2 \qquad (3)$$

Here L and v_∞ are constant.

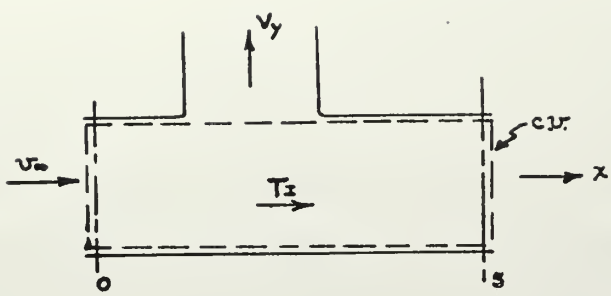
As the drift velocity decays from c_E to v_∞, clearly, by operating at as low as possible ejection velocity, c_E, the duration of the drift phase is minimized.

For a specified ejection velocity, c_E, and vessel speed, v_∞, the solution of the momentum equation (3) yields that the drift time is directly proportional to length, L, or for

$$c_D \longrightarrow 1.1 v_\infty, \quad t_D = \frac{3L}{2v_\infty} \qquad (4)$$

As the only means of transmitting thrust to the pulse jet during this phase is by wall friction, for this frictionless model the drift phase thrust will be zero.

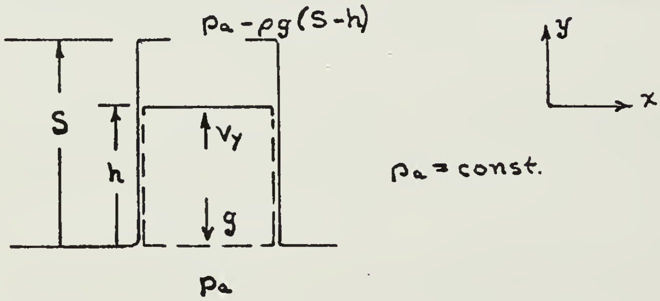
In our discussion of the inlet phase thrust, we shall consider only the axial portion of the pulse jet as no x direction momentum is changed within the chamber. Consider control volume, c.v..



A momentum balance in the x direction then is given by equation (5). (See Appendix A for derivation)

$$T_1 = 2 \rho A v_\infty v_y \tag{5}$$

Thus to find both the magnitude and duration of the inlet phase we must find the variation of chamber velocity, v_y, with time. Consider control volume, c.v..



Equation (6) is the resulting momentum balance between the time varying driving force and the inertia force.

$$h \frac{d^2h}{dt^2} = \rho g \ (s - h) \tag{6}$$

Note here that, as in the ejection phase, a nonlinear equation results.

Effect of Losses

We have determined the characteristics for an ideal (loss-less) pulse jet. However, realizing that we will have losses, we must now ask, "How do losses influence performance?" We shall consider here losses in the check valve, at the tee and those due to pipe and piston wall friction.

Without going into great detail, it appears that inlet time will be increased by all of the afore-mentioned losses. The output thrust derived during the ejection phase will be decreased by the piston wall friction, pipe friction and losses in the tee, and the duration of the drift phase will be decreased by pipe friction which will tend to decelerate the water.

In general, the more important losses appear to be those in the check valve, those at the tee, and those due to piston wall friction. Naturally, these effects tend to decrease the overall efficiency.

Gas Generation

To expel the working fluid, entrained sea water, the pulse jet requires a relatively high pressure fluid to be provided intermittently. This fluid will out of necessity be a gas, since we require a maximum volume displaced for a per unit weight of fuel consumed, and since, fortunately, many products of combustion are in the gaseous state even at elevated pressures. While it is recognized that there are many facets of the gas generator problem, those which we plan to discuss involve the energy storage material and gas generator characteristics as they are influenced by the characteristics of the idealized pulse jet.

The minimum weight requirement has limited our consideration to chemical fuels. The necessity for this limitation is due to the fact that at this stage in the development process, the fuel storage-gas generator equipment will most likely be a considerable fraction of the total plant weight because of the simplicity of the pulse jet.

The pulse jet requires intermittent ignition, while continuously operating expanders require ignition only on starting. This intermittent operation allows the use of solid fuels. Without stretching the imagination too far, we can foresee the possible use of a machine gun-like feeding mechanism for this application.

One characteristic of the pulse jet is that its fuel consumption will increase with depth. This is due to the increase in mass required to occupy a given chamber volume as pressure increases

and to the fact that the gas will not expand because of the small pressure ratio.

Many of the fuel identifying parameters for the pulse jet are similar to those for other deep operating propulsion systems. For any type of propulsion system, a liquid or solid fuel has an obvious advantage over gas because of its relatively high density. The possibility of cryogenic storage for a gaseous fuel and/or oxidizer should not be overlooked, however. Catterson and Swain note in reference (6) that liquid oxygen has a weight advantage to a depth of 20,000 feet in spite of the necessity for pressure hull containers.

The desirability of varying power output will influence the choice of a monopropellent or bipropellent (a solid fuel necessarily being a monopropellent). In a monopropellent one is not afforded the degree of control in varying the fuel-oxidizer ratio as in the case of the bipropellent.

The characteristics and state of the combustion products may have a marked influence on fuel selection. Solid products may foul the mechanism, while active products may interact with the water or chamber. The higher the combustion temperature, the greater the energy lost as heat and the lower the efficiency. Furthermore, the presence of products that are in the liquid state in equilibrium at environmental conditions, $40^{\circ}F$ and 10000 psi, may result in a loss in thrust per mass of gas generated due to condensing vapor.

Other factors should be considered in fuel selection, such as safety, ease of refueling, decomposition, and material

compatibility. However, they will not be discussed here as they are largely determined by specific mission requirements, and not by the specific requirements of the pulse jet.

Interface Phenomena

Interface phenomena may be summed up in one phrase: "losses, resulting from turbulence (or frothing) and from heat transfer."

Turbulence will be defined here as the degree to which the plane gas-liquid interface is disrupted. Clearly, the losses at the interface will increase as the useless random velocities of the water increase.

To determine the factors that control turbulence we must postulate how the random velocities are set up. During the pressure rise, the gas velocities experienced by the water surface are not exactly uniform across the plane surface. As the gas front will be spherical, the pressure exerted at the water surface will not be uniform; and some non-uniformities will be present in the gas front. The surface will become distorted, and this effect increases as the velocity perturbations become more prominent. For a pressure difference of about one atmosphere, a highly turbulent situation is foreseen.

To reduce the tendency toward turbulence, there appear to be four general approaches:

1. Turbulence can be reduced by properly distributing combustion, thereby providing for a uniform gas front to reach the entire water surface at the same instant, or by reducing the rate of pressure increase.

2. The effect of the turbulence can be reduced by providing an extra volume of water to buffer the turbulence.

3. The turbulence can be eliminated altogether by placing a solid piston at the interface, thereby doing away with it.

4. The turbulence can be reduced by reducing the surface area.

In an initial observation it would appear impossible (in practicality) to provide for a sufficiently uniform plane gas front. Furthermore, reducing the rate of pressure rise would be of little practical use as very little power would be developed. While some additional losses will certainly be unavoidable, the possibility of a buffering volume should be examined. However, the most feasible immediate solution appears to be eliminating the interface through the use of a piston.

Heat losses will depend upon the temperature gradient across the gas-liquid interface and the area of the interface. Thus, it would appear desirable to keep the combustion product temperature to a minimum. In keeping the contact area as small as possible, the length/diameter ratio should be made as large as possible and turbulence should be minimized.

Sea Water Exhaust Phenomenon

To investigate the influence of the exhaust phenomenon, we must review the work of R. Dernedde in reference (1). In his work at the Berlin Tank, Dernedde investigated in detail the hydrodynamics of this phenomenon. By mechanically driving a piston, he was able to impose sinusoidal motion on the entrained water and to determine the variation of efficiency with:

1. frequency
2. distance of the rear dead center of the stroke from the nozzle
3. rate of advance
4. stroke-diameter ratio

He found that no trend could be detected in his measurements when the frequency was changed by a factor of three and when the distance from the rear dead center to the nozzle was varied from 0 to 2 diameters. His results do show, however, that minimizing the stroke-diameter ratio to one would provide for maximum efficiency.

While these findings for a rear intake device are not directly applicable here, we can conclude in general that efficiency is a maximum for v/c of +1.0. Furthermore, the fact that the velocity of entrained water is greater than that of the stream provides for increased losses as a consequence of the useless kinetic energy of the relatively high speed ejected water.

CONCLUSIONS

While the drift phase model provides analytical results, the ejection and inlet phase considerations lead to nonlinear equations for entrained water length as functions of time. These expressions appear to be solvable only by numerical methods for each particular application. Therefore, the influence of the pulse parameters on its performance can only be ascertained by calculating the time average thrust, efficiency, and specific fuel consumption by varying each parameter and holding the others constant.

Thus, the usefulness of this model is somewhat limited. Even so, it is difficult to imagine a more simplified model that would at all represent satisfactorily the pulse jet cycle.

RECOMMENDATIONS

The findings of this study are in essence the mathematical model of the pulse jet. Of course, additional comments on its usefulness are in order.

The pulse jet cannot be recommended as a propulsive device for use in deep submergence applications when compared with present systems. Its use appears to be unfeasible for the following reasons:

Fuel consumption increases with depth;

The tendency for gas to expand at great depths is negligible;

The pulse jet provides only intermittent thrust;

Intermittent vertical thrusts are produced by changes in bouyancy and by vertical forces created during the cycle.

In shallow applications, the problems generated by depth are not present, and it is recommended that further efforts be undertaken with this in mind. Other difficulties may be reduced in magnitude if not eliminated by clustering and by staggering the firing sequence of pulse jets.

With these possibilities in mind, it is recommended that the experimental work proposed in the following sections be carried out to verify and/or refine the model presented.

EXPERIMENTAL PULSE JET

An attempt has been made to investigate the variation of the time-average-thrust of a pulse jet with its gas pressure, stroke, and the vessel speed. The pulse jet used in this study was designed to operate in the M.I.T. Propeller Tunnel, making use of the variable water speeds and strain gage dynamometer (See Figure VI, VII). Primary design considerations were:

1. attainment of constant gas pressure during the ejection phase,
2. simulation of the gas exhaust during the inlet phase,
3. elimination of the gas-liquid interface,
4. ease of installation in the tunnel,
5. ease of stroke setting,
6. light weight,
7. tolerable water flow into the vacuum system.

By running a 1/4 in. line from a 100 psig system, a first attempt was made to provide constant gas pressure. This proved unsatisfactory in that the pressure dropped significantly during the cycle and that it could not be varied. Better results were obtained by installing a surge tank near the pulse jet and by using a regulated pressure supply, thereby providing constant pressure down to 20 psig (Shown in Figure VIII).

Figure VI

ASSEMBLY DRAWING

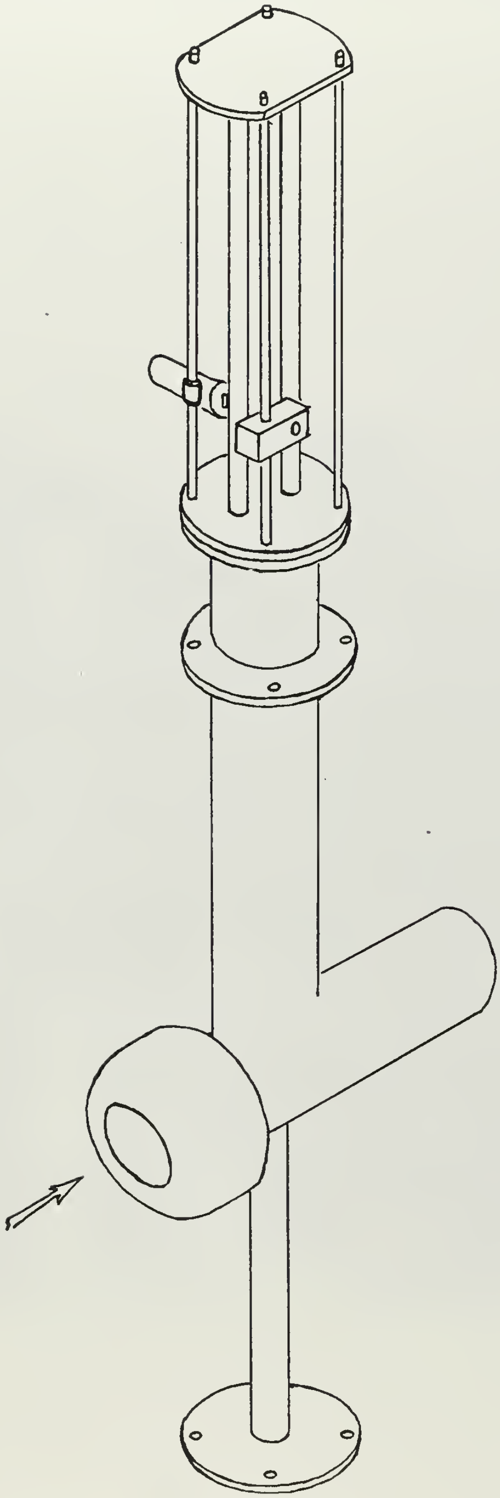


Figure VII

Pulse Jet Assembly Photograph

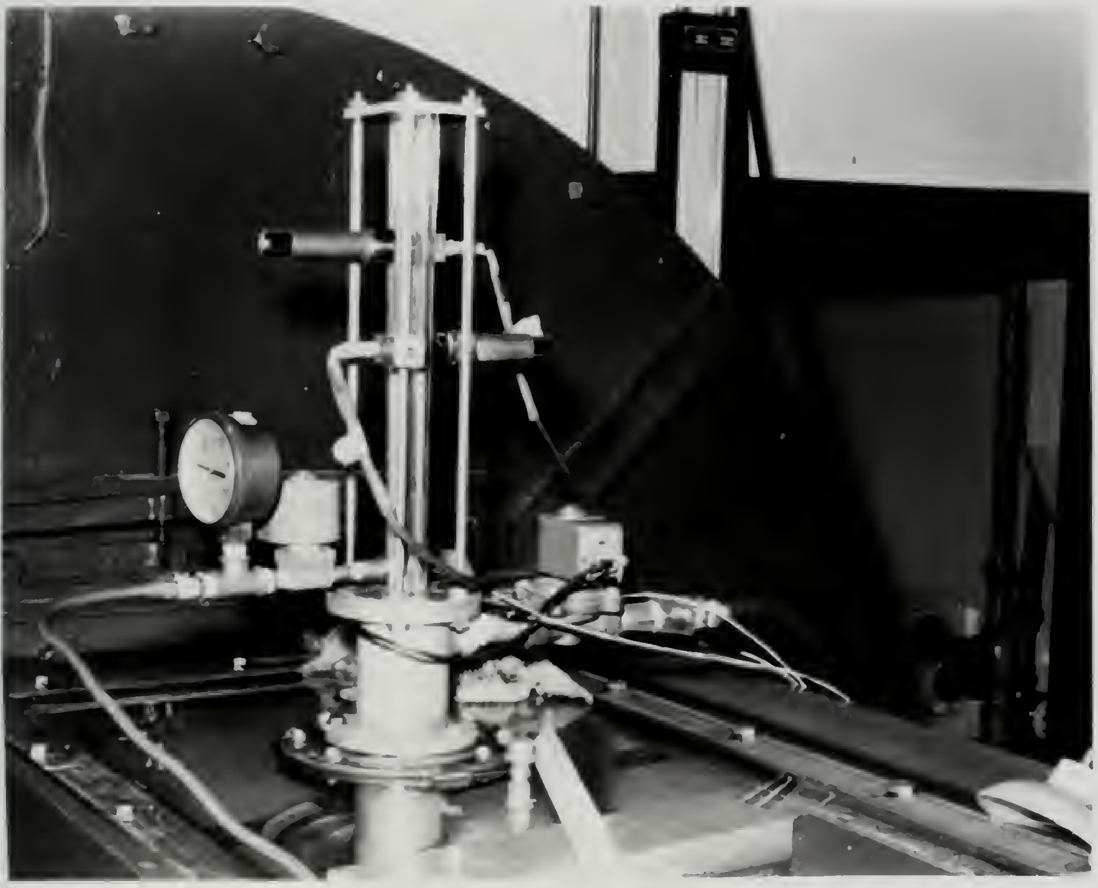
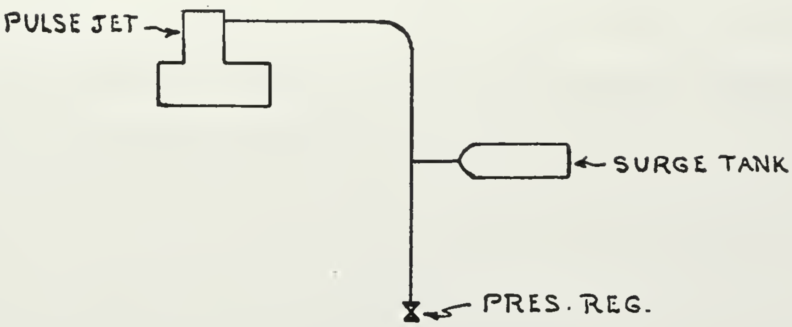


Figure VIII

Pressure System

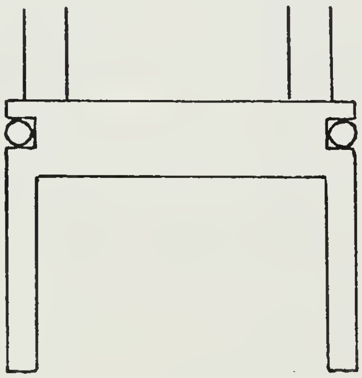


In simulating the gas exhaust, it was necessary to compensate for the stagnation conditions to which the gases in the test setup were being exhausted. In a moving vessel, gas would be exhausted to a medium with the same relative velocity as the inlet water. Here, however, the air is exhausted to the atmosphere with zero relative velocity. This velocity difference has been compensated for by exhausting the air into a vacuum.

In an effort to eliminate the interface while at the same time provide minimum wall friction drag, a piston with .005 in. clearance and no seal was installed. It proved unsatisfactory as an excessive amount of water passed by the piston. By inserting an O-ring in a similar piston, this problem was eliminated so that the pulse jet could operate; however, there was a substantial increase in drag (See Figure IX).

Figure IX

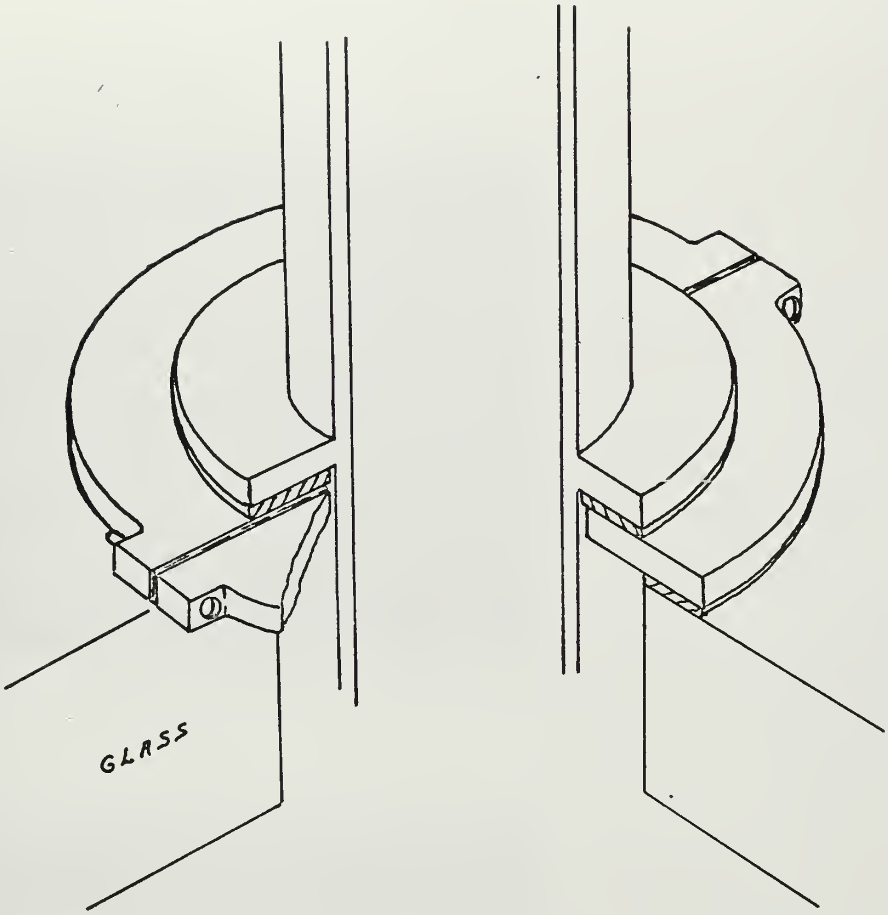
Piston Detail



The installation of the pulse jet in the tunnel is, by the necessity of its vertical mounting, a time consuming job. Windows with the mounting hole must be changed to vertical positions and the dynamometer secured at the bottom. The mounting and sealing arrangement at the top of the tunnel was satisfactory, although at times small leaks developed (See Figure X).

Figure (X)

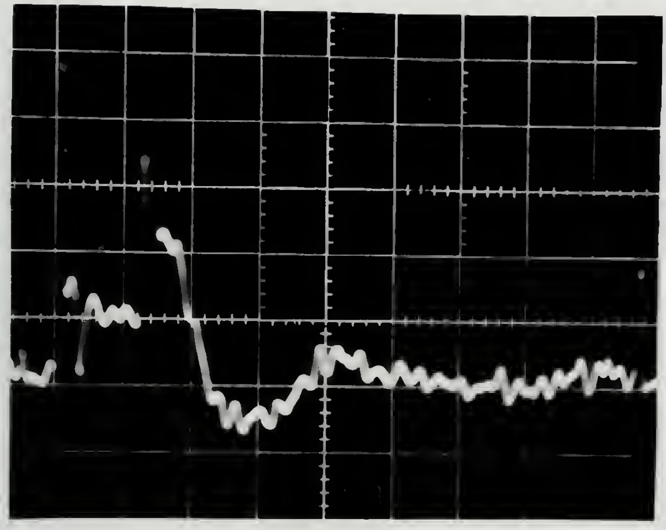
Mounting Detail



The arrangement for stroke variation also proved to be satisfactory. The battery-powered lights lasted over fifteen minutes. And when difficulty was encountered in triggering the photo-cells, it was easily corrected by making sure the lights were on or that they were not shining past the rod (For details, check Appendix C).

Figure XI

Experimental Results



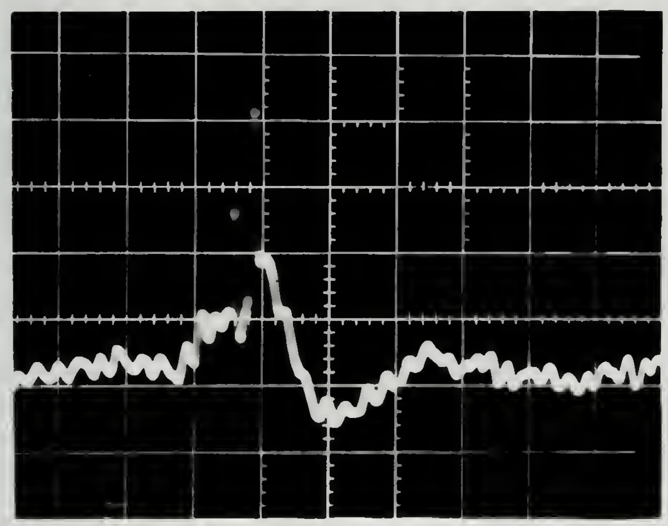
gas pressure - 100 psi

tunnel speed - 13.5 ft/sec

0-position + 3.5 cm

5 lb/cm

.1 sec/cm



gas pressure - 100 psi

tunnel speed - 19.5 ft/sec

0-position + 1.5 cm

10 lb/cm

.5 sec/cm

Discussion of Experimental Results, and Recommendations

The force curves obtained (Figure XII) proved to be unsatisfactory for a verification of an analytical analysis, because insufficient pressure regulation, (i.e., to below one atmosphere,), resulted in extreme overshoot. Regulation from 0 to 10 psi should provide sufficient ejection velocities for the tunnel speed limit, approx. 30 ft/sec. In addition, as a consequence of the tight piston seal, more force was required to move the piston than was provided by the head of water during the inlet phase. Even though operation was possible at a vacuum greater than that required by the velocity, the fact that an additional force had to be used made the inlet times obtained here of no consequence in the verification of the analysis.

Two alternatives to solving this latter problem appear to be either providing a looser seal, or using a different mechanism for gas exhaust.

Since little water was collected in the vacuum sump, it appears that a greater flow rate past the piston would be possible, and looser piston seal could be used*

A far more involved alternative would be to use a spring as a driving force during the inlet phase. This would require that the analysis be revised, which is a formidable task. However, in the longer run it may be proven that a spring return pulse jet is more satisfactory.

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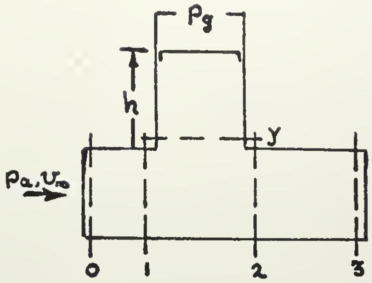
*It was necessary to drain the sump only at the end of a few hours of intermittent testing.

APPENDIX

APPENDIX A

Drift Phase



$$v_0 = v_1 \ , \ v_1 = v_2 + v_y$$

$$v_2 = v_3 \ , \ v_1 = v_2$$

$$p_3 = p_a \qquad p_g = \text{constant}$$

If the transient phenomena in section 1-y-2 and the effect of gravity in this section is neglected, we may write:

$$p_0 + \rho\frac{v_0^2}{2} = p_a + \rho\frac{v_\infty^2}{2} \tag{1}$$

and

$$p_1 + \rho\frac{v_1^2}{2} \overset{(2)}{=} p_y + \rho\frac{v_y^2}{2} \overset{(3)}{=} p_3 + \rho\frac{v_2^2}{2}$$

The momentum equations for each of the remaining sections are:

$$(p_y - p_g)A = \rho A h \frac{dv_y}{dt} + \rho g h A \tag{4}$$

$$(p_2 - p_3)A = \rho A L_{2-3}\frac{dv_2}{dt} \tag{5}$$

$$(p_0 - p_1)A = \rho A L_{0-1}\frac{dv_1}{dt} \tag{6}$$

with p_0, p_1, p_2, p_y, v_2, and v_y the unknown functions of time.

To find the resultant force on the pulse jet and the duration of this drift phase, we must solve for the velocities:

$$p_0 = p_a + \rho\frac{v_\infty^2}{2} - \rho\frac{v_1^2}{2} \qquad \text{from (1)}$$

$$p_1 = p_y + \rho\frac{v_y^2}{2} - \rho\frac{v_1^2}{2} \qquad \text{from (2)}$$

$$p_2 = p_y + \rho\frac{v_y^2}{2} - \rho\frac{v_2^2}{2} \qquad \text{from (3)}$$

$$p_y = p_g + \rho\bar{g}h\frac{dv_y}{dt} + \rho gh \qquad \text{from (4)}$$

Recall that $h = \int v_y dt$

then

$$p_y = p_g + \rho g\int v_y dt\,\frac{dv_y}{dt} + \rho g\int v_y dt \qquad (5)$$

The above equations for p_0, p_1, and p_2, rewritten in terms of v_2 and v_y, are:

$$p_0 = p_a + \rho\frac{v_\infty^2}{2} - \rho\frac{(v_2+v_y)^2}{2}$$

$$p_1 = p_g + \rho g\int v_y dt\,\frac{dv_y}{dt} + \rho g\int v_y dt + \rho\frac{v_y^2}{2} - \rho\frac{(v_2+v_y)^2}{2}$$

$$p_2 = p_g + \rho g\int v_y dt\,\frac{dv_y}{dt} + \rho g\int v_y dt + \rho\frac{v_y^2}{2} - \rho\frac{v_y^2}{2} \qquad (6)$$

The integrals may be eliminated by writing equation (6) in terms of h and v_2:

$$p_0 = p_a + \rho \frac{v_\infty^2}{2} - \rho \frac{(v_2 + \frac{dh}{dt})^2}{2}$$

$$p_1 \doteq p_g + \rho gh \frac{d^2h}{dt^2} + \rho gh + \rho/2 \left(\frac{dh}{dt}\right)^2 - \rho/2 \left(v_2 + \left(\frac{dh}{dt}\right)\right)^2$$

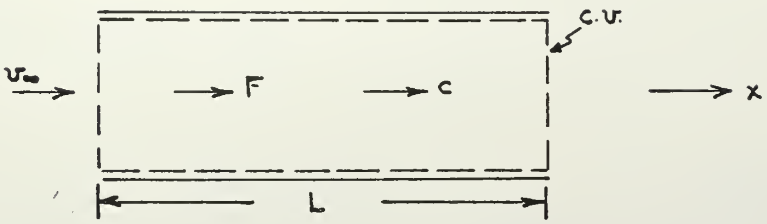
$$p_2 = p_g + \rho gh \frac{d^2h}{dt^2} + \rho/2 \left(\frac{dh}{dt}\right)^2 - \rho/2\; v_2^2 \qquad (7)$$

We have gained little here as equations (7) are very complicated, nonlinear differential equations. Thus it appears that our only alternative is to consider a simplified model with the hope that it will provide a straightforward yet applicable solution.

During the drift phase, the thrust on the pulse jet in the horizontal direction will be a drag force caused by the deceleration of the slug of water in section 0-3. The force will be of relatively small magnitude, but the duration for which it acts will be of great importance as it will be a considerable portion of the cycle time. Therefore we shall, for the moment, overlook the magnitude of this force and consdier its duration.

Some idea of the period of the drift phase may be attained by considering the time for a streamtube of relatively fast-moving water to decelerate.

Consider the control volume, c.v., below.



The momentum equation for this control volume, neglecting pipe friction, and for external forces equal to zero is

$$F_x = -\rho Acc + \rho Av\infty v\infty - \rho AL\frac{dc}{dt} = 0$$

or

$$L\frac{dc}{dt} + c^2 = v_\infty^2 \tag{8}$$

The solution for equation (8) is:

$$\int dt = \int \frac{Ldc}{(v_\infty^2 - c^2)} = \int \frac{Ldc}{(v\infty - c)(v\infty + c)}$$

$$t = \int \frac{Ddc}{(v\infty - c)} + \int \frac{Bdc}{(v\infty + c)}$$

where $L = (v\infty + c)A \pm (v\infty - c)B$

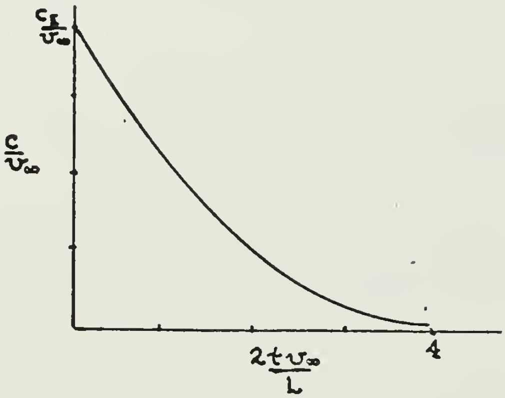
and $D = B = \frac{L}{2v\infty}$

then $t = \frac{L}{2v\infty} \log\left(\frac{v\infty + c}{v\infty - c}\right) + C$

At $t = 0$, $c = c_E = [2 \frac{(p_g - p_a)}{\rho}]^{1/2}$

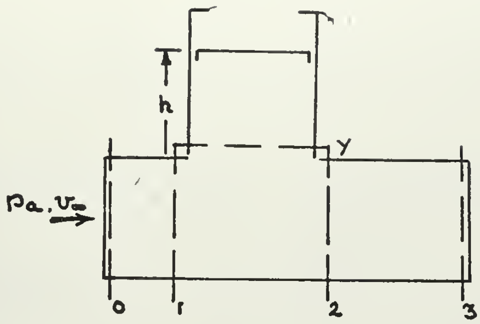
and $c = \frac{-L}{2v\infty} \log \left(\frac{v\infty + c_E}{v\infty - c_E}\right)$.

Finally $t = \frac{L}{2v\infty} \log \frac{\left(\frac{v\infty + c}{v\infty - c}\right)}{\left(\frac{v\infty + c_E}{v\infty - c_E}\right)}$ or $\frac{2tv\infty}{L} = \log \frac{\left(\frac{1+c/v\infty}{1-c/v\infty}\right)}{\left(\frac{1+c_E/v\infty}{1-c_E/v\infty}\right)}$



Now how can the drift time be minimized? Clearly operating at a c_E which will minimize $\log\frac{(\quad)}{(\quad)}$ will result in a minimum $\frac{2v\infty t}{L}$. This value of c_E is $c_E \longrightarrow v\infty$, making $\log \frac{(\quad)}{(\quad)} \longrightarrow 1$. For a minimum time given $\frac{2v\infty t}{L}$, we then require a minimum length, L, and a high vessel speed, $v\infty$. Finally, note that as pipe friction, the only vehicle for transmitting thrust to the pulse jet, has been neglected the drift phase thrust is of no consequence.

Inlet Phase



$$p_3 = p_a \quad p_a = \text{const.}$$

$$v_0 = v_1 , v_2 = v_3$$

$$v_1 = v_2 + v_y$$

By consideration of conditions in section 1-y-2, and by neglecting gravity the equations (1), (2), and (3) may again be written,

$$p_0 + \rho \frac{v_\infty^2}{2} = p_\infty + \rho \frac{v_\infty^2}{2} \tag{1}$$

$$p_1 + \rho \frac{v_1^2}{2} \overset{(2)}{=} p_y + \rho \frac{v_y^2}{2} \overset{(3)}{=} p_2 + \rho \frac{v_2^2}{2}$$

The changes in momentum in the remaining sections are given once again by equations (4), (5), and (6)

$$(p_0 - p_1)A = \rho A L_{0-1} \frac{dv_1}{dt} \tag{6}$$

$$(p_2 - p_3)A = \rho A L_{2-3} \frac{dv_2}{dt} \tag{5}$$

$$(p_y - p_a - \rho g h)A = \rho A h \frac{dv_y}{dt} \tag{4}$$

the unknowns being p_0, p_1, p_2, p_y, v_2, and v_y.

During this phase we require the internal drag force on the pulse-jet and the duration of the phase as determined by v_∞ and the stroke. This drag force equals $(p_3 - p_0)A$. We must now find this quantity in terms of the variables v_y and v_2.

$$(p_3-p_2)-(p_0-p_1) = -\rho L_{2-3}\frac{dv_2}{dt} - \rho L_{0-1}\frac{dv_1}{dt} \qquad (-5) + (-4)$$

$$p_3-p_0 = -\rho\left(L_{2-3}\frac{dv_2}{dt} + L_{0-1}\frac{dv_1}{dt}\right) + p_2 - p_1$$

$$p_2 - p_1 = \rho\frac{v_1^2}{2} - \rho\frac{v_2^2}{2} \qquad (2-3)$$

then

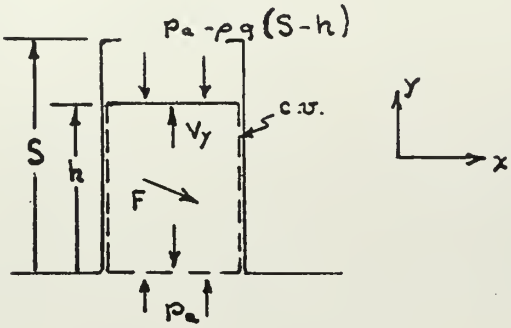
$$\frac{p_3 - p_0}{\rho} = \frac{v_1^2}{2} - \frac{v_2^2}{2} - L_{2-3}\frac{dv_2}{dt} - L_{0-1}\frac{dv_1}{dt}$$

and in terms of v_y and v_2

$$\frac{p_3 - p_0}{\rho} = \left(\frac{v_2 + v_y}{2}\right)^2 - \frac{v_2^2}{2} - L_{2-3}\frac{dv_2}{dt} - L_{0-1}\frac{d(v_2+v_y)}{dt}$$

Thus a solution for v_y and v_2 will yield the internal drag force and its duration. But before proceeding, note that the expressions for v_2 and v_y are identical to those of the drift phase, complicated non-linear

equations. Without attempting an exact solution, we shall try to simplify our model by calling $p_y = p_a$ = constant.



The momentum equation for c.v., in vertical direction, y, is

$$F_y = \rho A v_y v_y - \frac{d}{dt}(\rho A h v_y)$$

$$= \rho A v_y^2 - \rho A v_y \frac{dh}{dt} - \rho A h \frac{dv_y}{dt}$$

and as $v_y = \dfrac{dh}{dt}$

$$F_y = -\rho A h \frac{dv_y}{dt}$$

The external forces on this control volume are given by equation (9)

$$-F_y = (p_a - p_a - \rho g(s-h))A \tag{9}$$

The inertia force must equal the external forces on the control volume, or

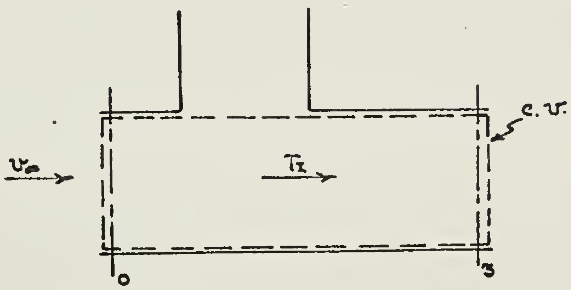
$$\rho A h \frac{dv_y}{dt} = \rho g A(s-h)$$

Recall that $h = \int v_y dt$.

Then $\frac{hd^2h}{dt^2} = g(s-h)$

The only means of solving this nonlinear equation would appear to be by numerical techniques.

To determine the inlet phase thrust, consider the control volume, c.v.



Here we shall assume the effects of increase in velocity upstream and the decrease in velocity downstream to balance out and the resultant thrust to be negligible. The only thrust then will be a negative force due to the deceleration of the mass taken into the chamber, or

$$T_I = F_x = -\rho A v_3^2 + \rho A v_0^2 + \frac{d}{dt}(\rho A L_{0-3} v_x)$$

and as $\frac{d}{dt}(\rho A L_{0-3} v_x) = 0$

$$T_I = -\rho A(v_0 - v_y)^2 + \rho A v_0^2 = \rho A(2v_0 v_y - v_y^2). \qquad (10)$$

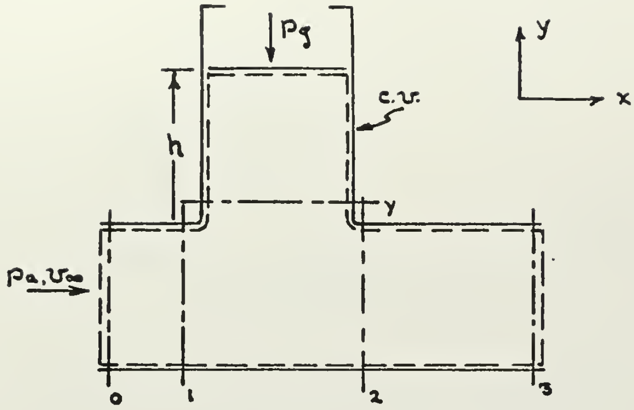
If we also assume, $v\infty = \frac{v_0 + v_3}{2} = \frac{2v_0 - v_y}{2}$ and

$$2v_0 = 2v\infty + v_y, \qquad (11)$$

by substitution of equation (11) into equation (10), we find that

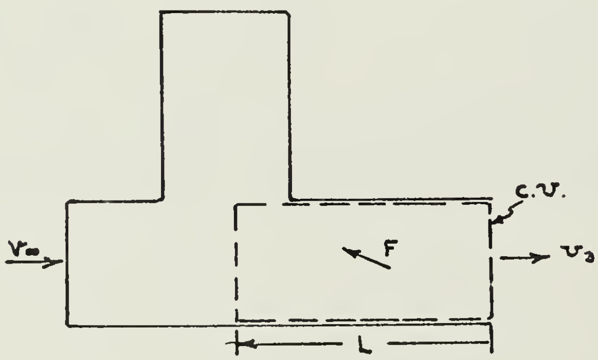
$$T_I = \rho A(2v\infty v_y + v_y^2 - v_y^2) = 2\rho A v\infty v_y.$$

Ejection Phase



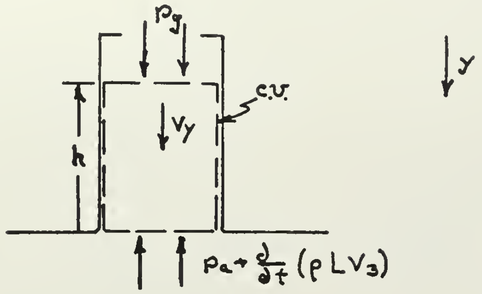
To determine the forces in the horizontal and vertical directions during the ejection phase, we must write the momentum equation in these directions for the control volume, c.v., above.

When the x direction is considered, momentum is changed in the control volume shown below.



and $F_x = -\rho A v_3 v_3 - \frac{d}{dt}(\rho A L v_3)$.

In the y direction, neglecting gravity, momentum is changed in vertical control volume, c.v.,



$$\text{and } F_y = -\rho A v_y v_y - \frac{d}{dt}(\rho A h v_y) = -\rho A h \frac{dv_y}{dt} .$$

To find F_x, we must find v_3. This can be accomplished by equating the external forces in the y direction to F_y, or $F_y = (p_g - p_a)A - \rho AL \frac{dv_3}{dt}$, neglecting the mass of the piston and friction at the walls.

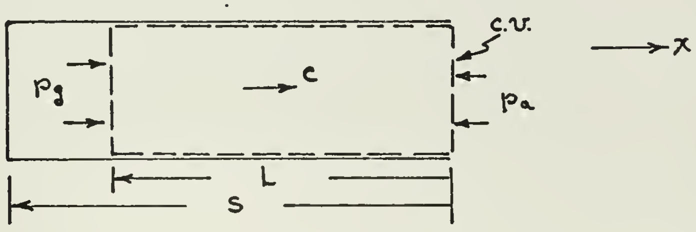
Now

$$p_g - p_a - \rho L \frac{dv_3}{dt} = \rho h \frac{dv_y}{dt} , \text{ and as } v_y = v_3$$

$$\frac{p_g - p_a}{\rho} = (h+L)\frac{dv_3}{dt} .$$

As this equation is nonlinear, we shall avoid its solution by considering a simplified model. Hopefully, it will provide a sufficiently realistic solution.

Consider the control volume, c.v.,



Then the x direction momentum is given by equation (12).

$$F_x = - \rho Acc - \frac{d}{dt}(\rho ALc) \tag{12}$$

Observe that $L = s - \int cdt$

and

$$\begin{aligned} F_x &= - \rho Acc + \rho Acc - \rho AL \frac{dc}{dt} \\ &= - \rho AL \frac{dc}{dt} \end{aligned} \tag{13}$$

Equation (13) rewritten in terms of L is

$$F_x = - \rho AL \frac{d^2L}{dt^2}$$

Now if we equate F_x to the external force, we find that

$$p_g - \frac{p_a}{\rho} = -L \frac{d^2L}{dt^2}$$

It would appear that the only means of solution to this nonlinear equation would be by numerical methods.

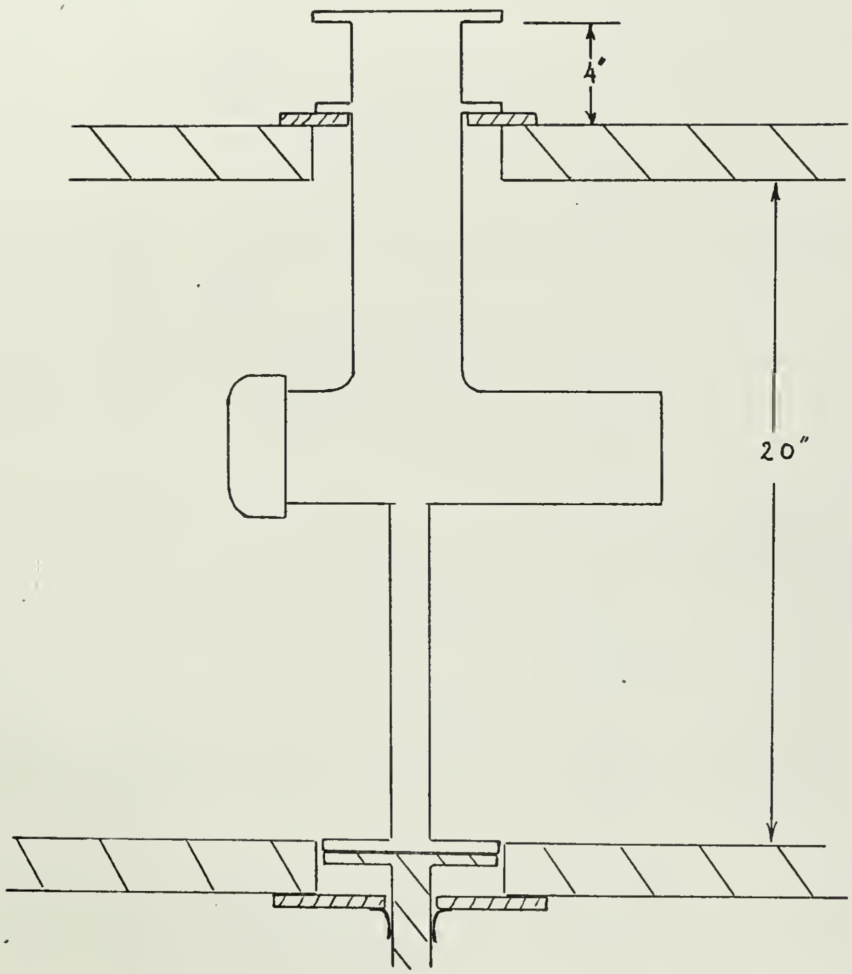
APPENDIX B

Experimental Setup

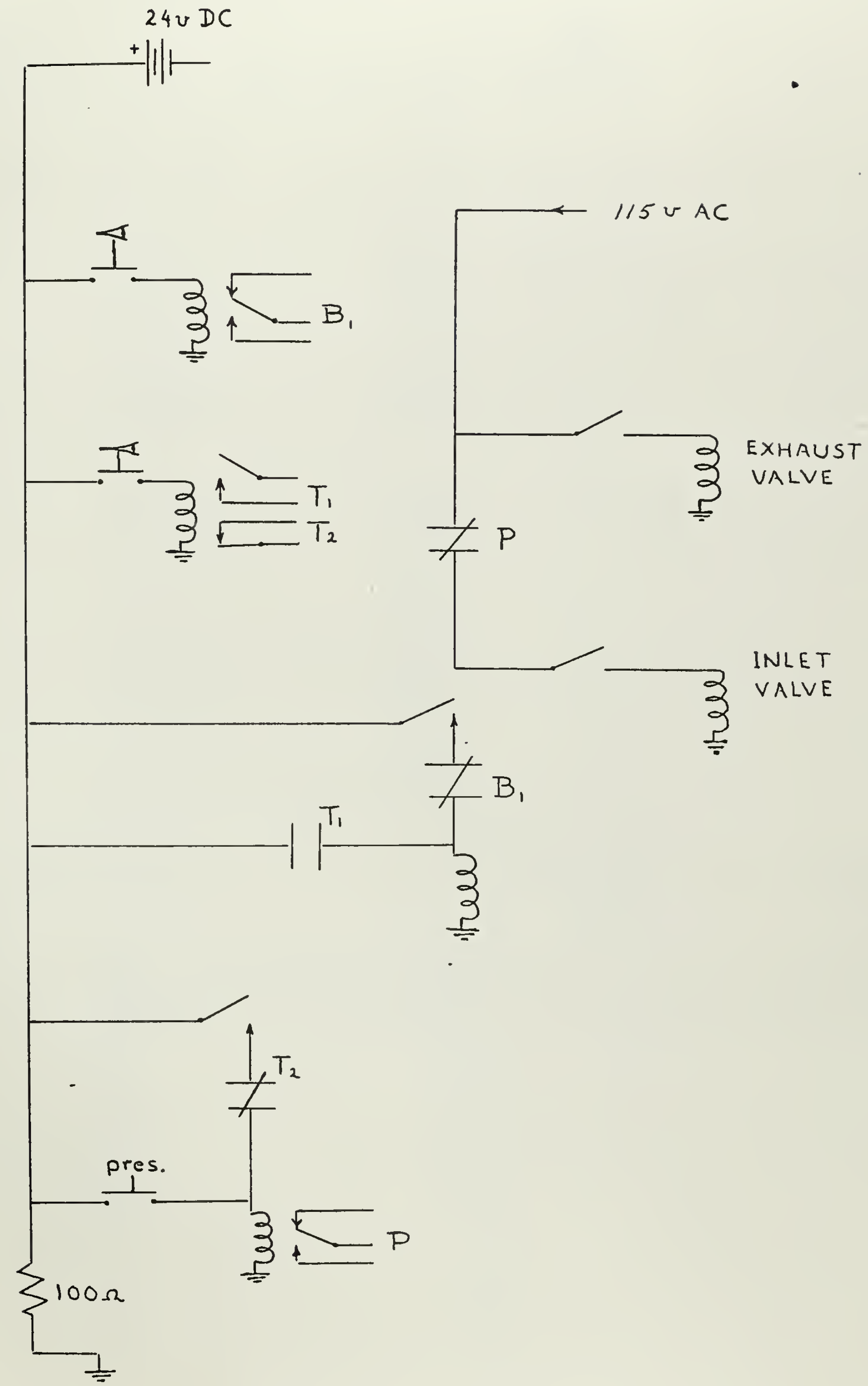
Force traces were recorded for various gas pressures, stroke settings, and vessel speeds by presenting the signal of a (Budd Model P-350) strain indicator, connected to the upper set of gages of the installed dynamometer on a Tektronix Type 502 oscilloscope.

APPENDIX G

INSTALLATION DIAGRAM

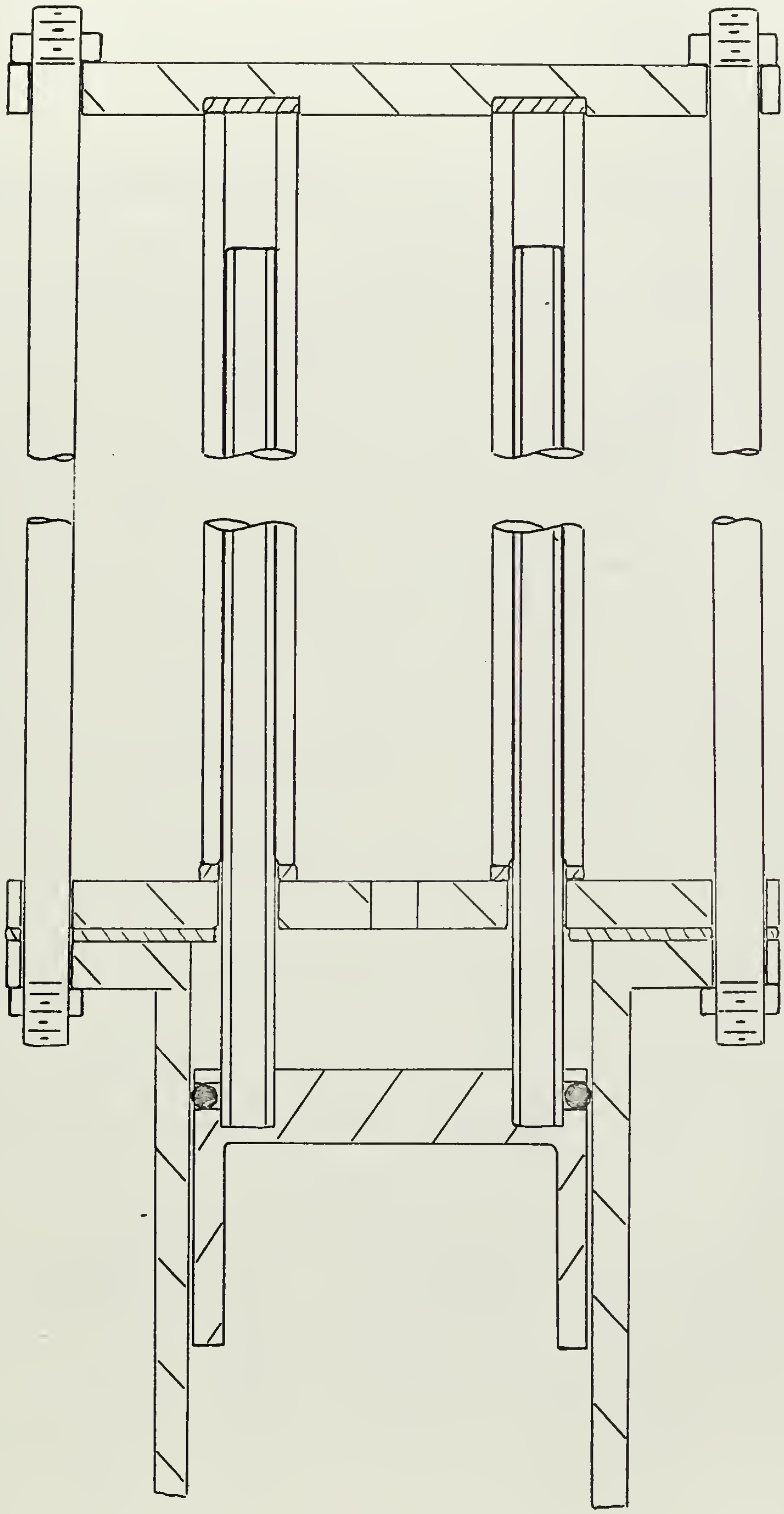


CONTROL CIRCUIT

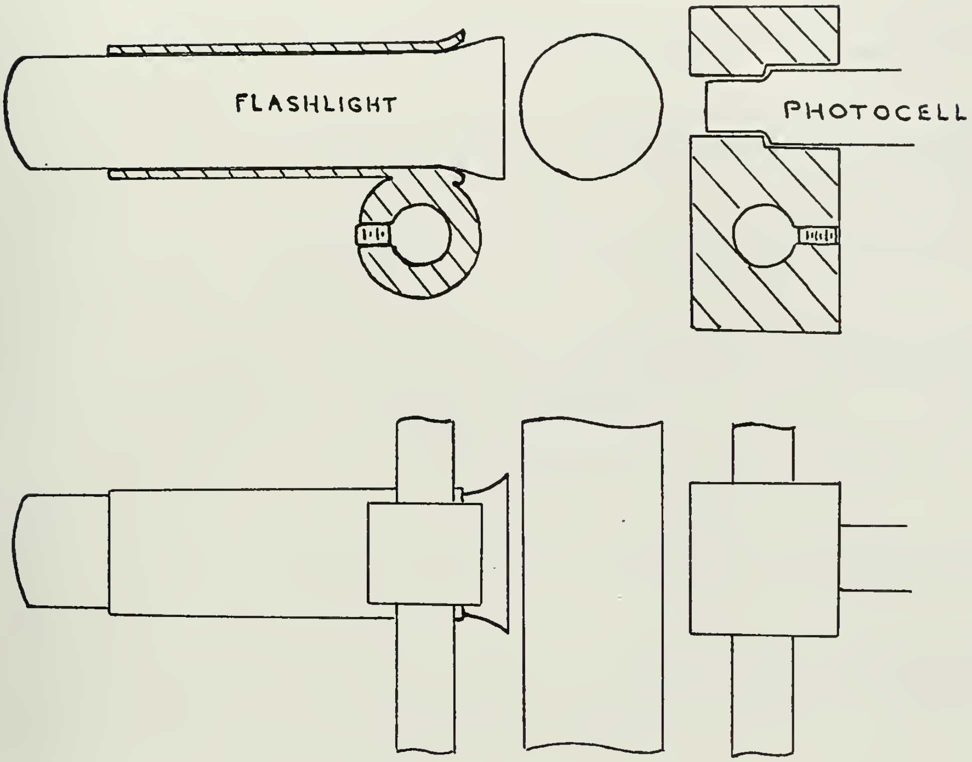


Piston & Control Rod

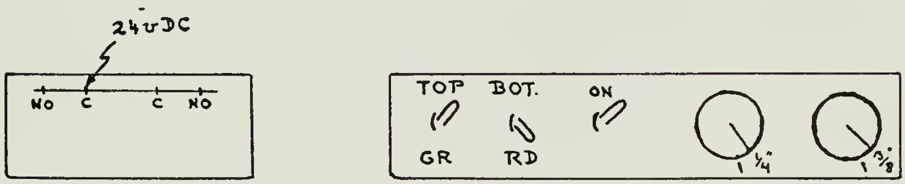
Detail



PHOTOCELL DETAILS



CONTROL PANEL SETTINGS



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